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THERMODYNAMIC EVALUATION OF SHOCKLESS EXPLOSION COMBUSTION FOR GAS TURBINE POWER CYCLES

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ABSTRACT

The efficiency of gas turbine power plants has experienced a steady increase in the last decades. Novel cooling technologies and materials allowed higher turbine inlet temperatures and led to higher system efficiencies. However, this trend has an upper limit due various cost and environmental factors.

A way to bypass this problem would be to search for alternative system configurations and novel cycles. Constant volume combustion (CVC) cycles could be such a solution, if several challenges could be addressed and solved. Two constant volume cycles are relevant in most thermodynamic studies, namely the Humphrey and the ZND cycle. While the first considers purely constant volume combustion, the second refers to the use of detonations as a way to approach CVC. Numerous thermodynamic investigations have been performed in the past on both cycles. Nevertheless, most studies are limited to simple assessments of the ideal cycle efficiencies, without taking into account the characteristics of an actual system operating on them.

Aim of the current work is a comparison of the Humphrey (model of shockless explosion combustion) cycle to the typical Joule cycle. A quasi steady state approach is used to evaluate the thermodynamic performance of the SEC cycle. The efficiency of the turbine in the respective unsteady inflow conditions is evaluated with steady state performance maps.

INTRODUCTION

The potential of constant volume combustion to improve the performance of gas turbines and propulsion systems has motivated many research projects in the past. Initially, research has concentrated primarily on the realization of combustion chambers to accommodate repeatable detonations or even a newer concept of rotating detonation waves[1], [2]. Apart from these methods, Bobusch *et al.* have introduced an alternative constant volume combustion method called shockless explosion combustion (SEC)[3].

Despite the numerous studies on constant volume combustion, the fundamental thermodynamic evaluation of possible cycles operating with CVC remains an open issue. The first studies, a thorough summary of which can be found in the work of Coleman[4], concentrated on propulsion systems operating on the ZND or the Humphrey cycle. Heiser and Pratt [5] on the other hand focussed on the basic processes of the thermodynamic cycle and showed that the isentropic efficiency of the expansion process is crucial for the performance of CVC cycles for power generation.

As a result many succeeding projects, like that of Schauer *et al.* [6], studied the interaction of turbines with a detonation tube connected directly at their inlet. Rasheed *et al.* [7] performed such a study with a multi-tube pulse detonation combustor, whereas Rouser *et al.*[8]–[10] concentrated on the measurement of the actual unsteady response of a radial inflow turbine. Paxon and Kaemming [11] introduced the concept of turbine quasi-transient modelling and computed the expansion work in a CVC period by mass integration.

The current work uses the Humphrey cycle as a simplified model to compute the performance of the shockless explosion cycle. A short introduction to the combustion process is followed by the presentation of the implemented model. The work concludes with a performance comparison of the SEC cycle to the Joule cycle.

SHOCKLESS EXPLOSION COMBUSTION

Shockless explosion combustion is a cyclic constant volume combustion process that makes use of a standing wave in a combustion chamber [12]. It consists of four stages, depicted in Figure 1. The first stage starts at the time when the wave reaches the closed end of the combustion tube and is reflected on it. The low pressure behind the wave is used to fill the combustor with air. During the second stage, fuel is injected in the tube. The equivalence ratio is controlled in such a way that the ignition delay in the mixture matches its residence time and the oscillation period of the standing wave. This means that the fuel injected first, which stays the longer in the combustor, must have the longer ignition delay time and thus the mixture it forms with air is the leanest. By the same token, fuel injected last must form a richer mixture with air. The total injection time is equal to the time the wave needs to reach the open combustor end and return as a pressure wave to the closed end. This is also defined as the third phase of the SEC process. Just before the pressure wave reaches the closed end, the entire fuel-air volume undergoes homogeneous auto-ignition and is burnt without any shock waves. Moreover, combustion occurs at the same time when the standing wave is raising the pressure at the tube inlet. The wave is thus amplified and has enough energy to get reflected as a suction wave when it reaches the closed end of the tube and restart the process.

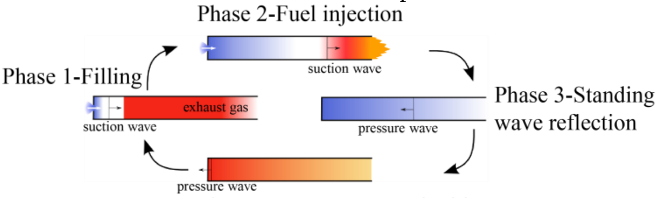

Figure 1 Shockless explosion process

Several studies of the SEC, both experimental and theoretical, have been carried out in the scope of the German collaborative research centre 1029. Berndt *et al.* [12]–[14] concentrated on the mathematical and chemical kinetic modelling of the process. They used the 1-D unsteady and compressible Euler equations to model the relevant gas dynamics. Combustion reactions were modelled with appropriate chemical kinetics and appeared in the equations as source terms. Reichel *et al.* [15] presented an experimental study of the process under atmospheric conditions and with Dimethyl ether (DME) as a fuel. Their main aim was to develop a model based control algorithm that would allow them to control the fuel injection pattern. Their results showed that the desired fuel stratification could be achieved achieved, thus leading to a significant reduction in spatial variance of the auto-ignition delay times.

MODELLING METHODOLOGY

The Humphrey cycle is chosen to model the SEC cycle, whereas the Joule cycle is used to represent conventional gas turbine cycles. The T-s and P-v diagrams for the ideal cycles for the same turbine inlet temperature are presented in Figure 2. Both cycles start with an isentropic compression, followed from the combustion process. In the case of the Joule cycle this process is isobaric, whereas for the SEC cycle it is isochoric. An isentropic expansion process follows and the cycle closes with an isobaric cooling.

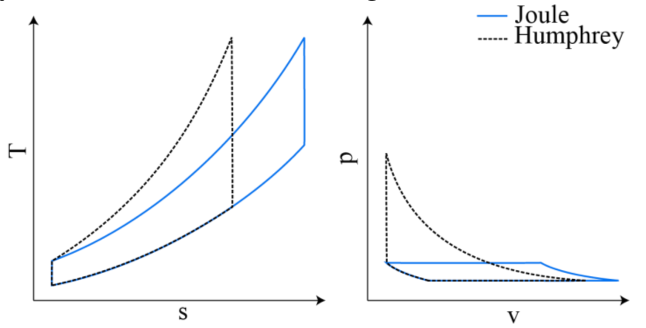

Figure 2. T-s and p-v diagrams of the ideal cycles

A central issue in modelling non-ideal CVC cycles is their inherent dynamic nature, which cannot be sufficiently reproduced by steady state thermodynamic models. In the current work, it is assumed that the unsteady operation of the combustion chamber has no influence on the operation of the compressor. The compression process is modelled with the same constant polytropic efficiency, equal to 0.9, independent of the compression ratio and the mass flow rate of air. The outlet temperature and the mass specific work consumtion of the compressor are given from equations (1) and (2) .

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\eta_{p,C}\cdot\gamma}} \qquad (1)$$

$$w_C = \frac{\gamma}{\gamma-1}\cdot RT_1\cdot\left(\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\eta_{p,C}\cdot\gamma}} - 1\right) \qquad (2)$$

For the Joule cycle the combustion chamber is modelled by a simple isobaric heat addition. The temperature at its exit is calculated by solving the nonlinear energy conservation equation. Shockless explosion combustion, being a cyclic combustion process, results in outlet gas conditions that are a function of time. In order to reduce the computational time and complexity of the SEC model, the simplified model presented from Endo and Fujiwara [16] has been implemented and adapted. Polytropic gases and one-dimensional flow inside the combustion tube are assumed and the combustion process is divided in the filling, the combustion and the exhaust phases. This simple lump sum model ignores the details of the described gas dynamic phenomena, and is depicted in Figure 3. At point t1, heat is added spontaneously to the working medium under constant volume. The conditions at the chamber inlet are taken as starting conditions for the constant volume heat addition to calculate the state at the end of this process. The temperature after isochoric combustion is calculated by solving the nonlinear equation for conservation of energy.The blowdown process (phases t2 and t3) is modelled as an isentropic mass extraction process from a volume at uniform pressure and temperature. This two-step process results in the time dependent values of mass flow, pressure and temperature at the inlet of the turbine. The pressure inside the combustion chamber at four time steps is presented qualitatively in Figure 3.

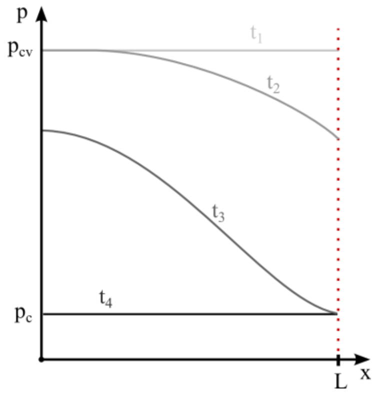

Figure 3. Simplified model of the SEC process.

The time series of temperature and pressure at the combustion chamber outlet are directly used as turbine inlet conditions. The expansion is modelled with a typical turbine performance map, presented in Figure 4, where the turbine isentropic efficiency is expressed as function of its expansion ratio. This function results from a typical turbine map [17]–[19], whereas in the present study the turbine is assumed to rotate at a constant speed. Based on the pressure of each exhaust gas mass increment, the corresponding turbine efficiency is computed and the expansion of the mass increment is performed with this value. The total work generated from the turbine in a limit cycle results from the integration of the work produced by each mass increment according to eq.(3)

$$w_T = \frac{1}{\int_0^1 dm_{T,in}} \int_0^1 \eta_{is,T} c_p T_{T,in} \cdot \left(1 - \Pi_T^{-\frac{\gamma-1}{\gamma}}\right) dm_{T,in} \quad (3)$$

The cycle work output is then computed from the difference between the compressor work consumption and the turbine work production. The efficiency the cycle is given from eq.(4).

$$\eta = \frac{(m_f + m_a) w_T - m_a w_C}{m_f \cdot h_{PR}} \quad (4)$$

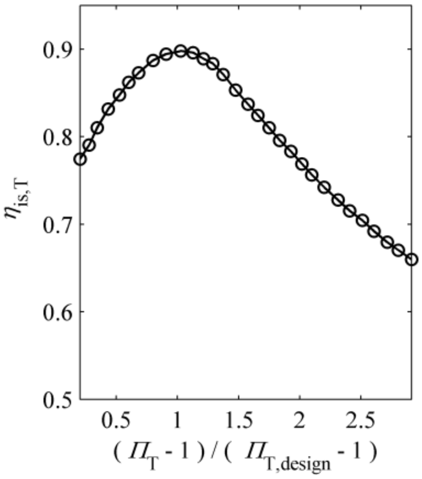

Figure 4. Simplified turbine performance map

Machines operating on the Joule cycle are optimized, real systems that represent the state of the art. In the contrary, no commercial gas turbines that operate with the SEC cycle exist, and thus the comparison must be kept on a conceptual level. This comparison can be carried out either for the same dimensionless specific fuel heat input or for the same mass average turbine inlet temperature (TIT). Both methods are used in the current work, whereas the second is carried out at 950°C and 1500 °C. The first value is representative for modern micro gas turbines, whereas the second refers to modern heavy duty gas turbines.

$$\tilde{q} = \frac{f \cdot h_{PR}}{c_p \cdot T_0} \quad (5)$$

The dimensionless specific fuel heat input is defined by Eq.(1) and is a function of the equivalence ratio of combustion and the properties of the combustion mixture. Although the TIT computation is quite straightforward for the Joule cycle, this is not the case for the SEC cycle, due to its inherent unsteadiness. If one decides to use the maximum temperature value of the combustion process as the TIT, the result will underestimate the efficiency of the SEC cycle. This maximum temperature occurs for a very short time interval and applies only for a very small exhaust mass increment. For this reason the mass averaged temperature has been used, defined in eq.(6). This value is also representative for the necessary cooling air flow, so that one could assume that the respective losses from the secondary air systems would be comparable in SEC and Joule cycles and thus ignored from the current study.

$$TIT_{m,av} = \frac{1}{\int_0^1 dm_{T,in}} \int_0^1 T_{T,in} dm_{T,in} \quad (6)$$

Apart from the choice of TIT, the way that the turbine map is chosen also plays a crucial role for the aspired comparison. Each map has an optimum expansion ratio and the efficiency of the expansion process drops for lower or higher ratios. For the Joule cycle the definition of the optimum ratio of the turbine is straight forward. In the SEC cycle the varying pressure at the combustion chamber outlet complicates this choice. In the current work, the turbine design ratio was computed with an optimization algorithm and was chosen equal to that at which the maximum efficiency of the whole cycle occurred. Since the outcome of the SEC process depends on the mixture properties, this value was computed every time the inlet conditions of the combustion chamber changed.

RESULTS AND DISCUSSION

Figure 5 presents the thermal efficiency of the analyzed cycles as a function of the compressor pressure ratio and for TIT values of 950°C 1500°C respectively.

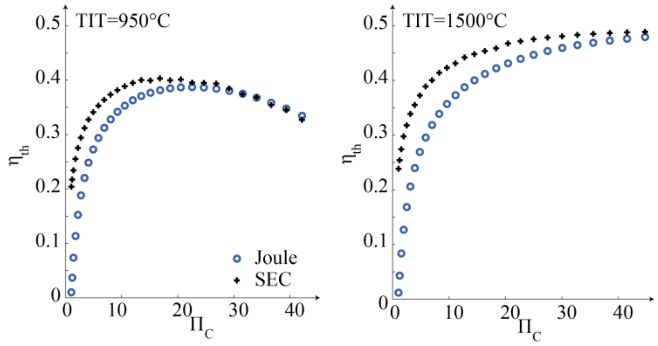

Figure 5 Thermal efficiency of the cycles

For both TIT values and for relatively low compressor pressure ratios the efficiency of SEC cycle is higher by 30-50% than that of the Joule cycle. The SEC cycle loses its advantage against the Joule cycle for pressure ratios above 20 for a TIT of 950°C and above 40 for a TIT of 1500 °C. This is due to the already mentioned sensibility of their efficiency on the turbine isentropic efficiency.

The comparison of the cycles for a constant dimensionles fuel thermal energy input has an equivalent outcome, as can be seen in Figure 6. Comparing this figure with the results of Heiser and Pratt [5], one can also state that the use of the current model reveals a similar behaviour for the cycles.

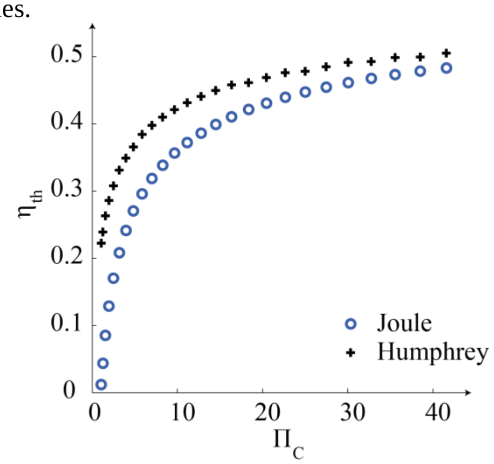

Figure 6. Thermal efficiency of the cycles for dimensionles fuel energy input $\tilde{q} = 5.4$

The SEC cycle efficiencies are computed with the turbine optimum design expansion ratio. This ratio was computed with an iterative process by changing its value and calculating the respective turbine work extraction for a given time series of the combustion chamber exhaust conditions. Figure 7 shows the values of the optimum expansion ratio of the SEC cycle, for the cases presented in Figure 5 and in Figure 6, as a function of the compressor pressure ratio.

The optimum design expansion ratio of the SEC cycle (and in general the CVC cycles) is a function of both the compressor pressure ratio and the TIT values. In the cycles with a TIT value of 1500°C the optimum design expansion ratio increases with a higher rate than the compression ratio up to a Π_C value of approximately 20 and then their increase rate is equivalent. A similar behavior is observed for the cycles with a TIT value of 950°C, but this change in the increase rate takes place at a lower compression ratio. Furthermore, higher values of the TIT result in higher turbine design expansion ratios. For example, turbine of a SEC cycle with a TIT of 1500°C and a compressor pressure ratio of 10 must be designed for a nominal expansion ratio of approximately 20.

These findings are a result of the direct connection between the outlet conditions of pressure SEC combustion chambers to the dimensionles fuel thermal energy input and their inlet pressure. In order to have the same TIT value for increasing compression ratios, the combustion equivalence ratio must be adapted respectively. Its values for the presented simulations are shown in Figure 8. At the same time, the equivalence ratio of the combustion process is directly connected to the dimensionles fuel thermal energy input, as can be seen in Table 1. While these changes have no impact on the turbine inlet pressure of the Joule cycle, lower equivalence ratios lead to a lower design turbine expansion ratios of the SEC cycle.

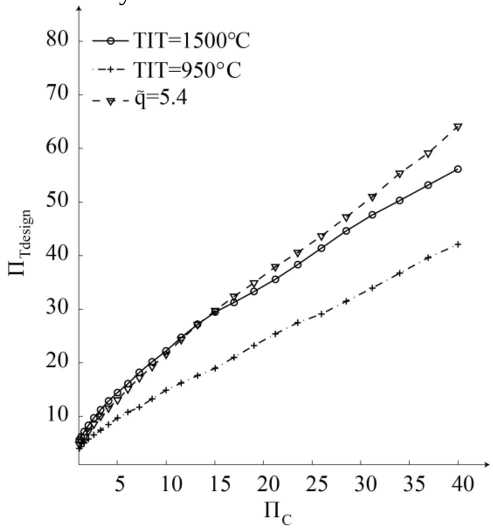

Figure 7. Optimum turbine design expansion ratio in a SEC cycle

Table 1. Equivalence ratio as a function of the combustion specific heat addition

Φ	1	0.67	0.5	0.4	0.33	0.29	0.25	0.22	0.2
$\tilde{q}$	10.8	7.20	5.4	4.3	3.6	3.1	2.7	2.4	2.2

Besides, the equivalence ratio of the combustion process is relevant for its feasibility. Combustion of lean mixtures at constant pressure is the state of the art method for low emissions and high combustion efficiency in gas turbines. By contrast, it is hardly possible to realize detonations of lean mixtures with a reasonable deflagration to detonation transition length [20]. On the other hand, constant volume combustion with the SEC process could be also possible for globally lean mixtures [3].

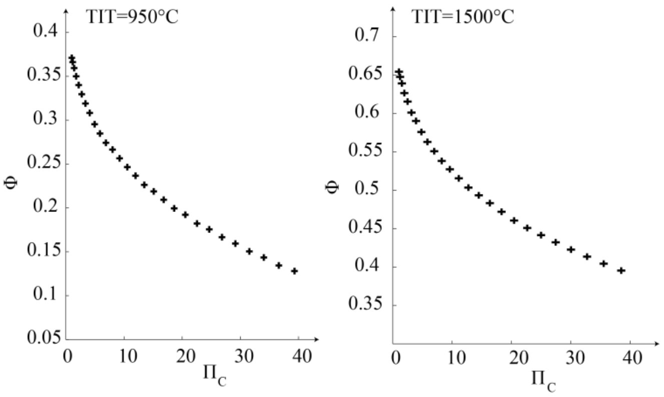

Figure 8. Equivalence ratio as a function of the compressor pressure ratio a given TIT

CONCLUSIONS

The current study explored some of the assumptions of the thermodynamic simulations of the SEC cycle. It has been shown that under certain conditions constant volume combustion and specifically SEC can significantly increase the efficiency of gas turbines. Especially for low compression ratios, the SEC cycles can achieve efficiency values that are by even 20% higher than these of the Joule cycle. In contrast, its efficiency advantage vanishes for higher compression ratios.

In conclusion, further detailed studies are needed in order to clarify the role of transient and gas dynamic phenomena in the operation of gas turbines that operate on CVC cycles and the SEC cycle. The simplified turbine model used in the current study will be replaced in the future by a high fidelity loss model. Turbines belonging to CVC engines would be forced to operate in a very broad values spectrum of inlet velocity and pressure. Consequently novel loss models are necessary in order to account for the respective additional shock losses at the inlet of the turbines and the losses for the frequent off-design operation.

NOMENCLATURE

Abbreviations

CVC: Constant Volume Combustion
SEC: Shockless Explosion Combustion
TIT: Turbine Inlet temperature
ZND: Zeldovic, von Neuman , Döring

Latin Characters

c_p: Heat capacity ratio [J/kgK]
h_{PR}: Lower heat value [J/kg]
f: Combustion fuel to air ratio
m: Mass flow rate [kg/s]
p: Pressure [bar]
p_c: Pressure at the outlet of the compressor [bar]
p_{cv}: Pressure resulting from CVC [bar]
$\tilde{q}$: Dimensionless fuel thermal energy input [-]
s: Specific entropy [J/kgK]
R: Universal gas constant [J/molK]
T: Temperature [°C]
v: Volume [m^3]
w: Mass specific work [J/kg]

Greek Characters

γ: Heat capacity ratio[-]
η_{th}: Cycle thermal efficiency [%]
$\eta_{p,C}$: Compressor polytropic efficiency [%]
$\eta_{is,T}$: Turbine isentropic efficiency [%]
Π_C: Compressor pressure ratio [-]
Π_T: Turbine expansion ratio [-]
$\Pi_{T,design}$: Design turbine expansion ratio [-]
Φ: Equivalence ratio of the combustion reaction [-]

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